AD-A133 072 FLUIDIZED BED AIR-TO-AIR HEAT PUMP EVAPORATOR EVALUATION(U) NAVAL CIVIL ENGINEERING LAB PORT HUENEME CA J L ASHLEY JUL 83 NCEL-TN-1673 1/1

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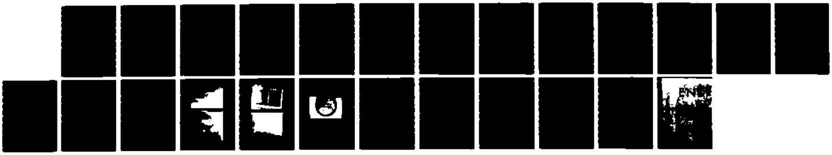

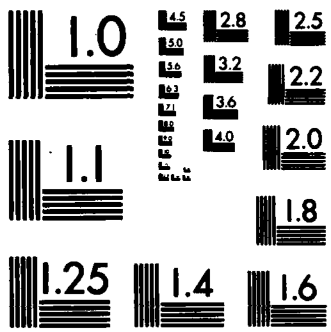


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# TECHNICAL NOTE

TN NO: N-1673

TITLE: FLUIDIZED BED AIR-TO-AIR HEAT PUMP EVAPORATOR EVALUATION

AUTHOR: Joseph L. Ashley

DATE: July 1983

SPONSOR: Naval Material Command

PROGRAM NO: ZF66-512-001-095

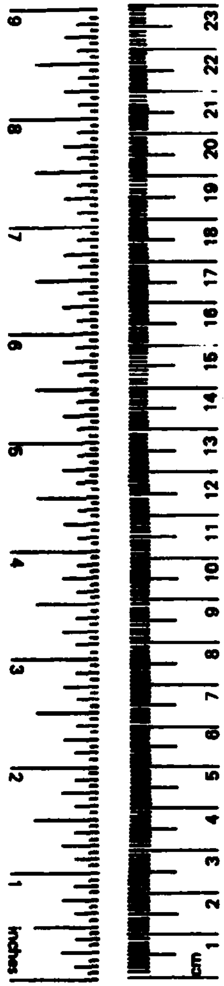
NAVAL CIVIL ENGINEERING LABORATORY
PORT HUENEME, CALIFORNIA 93043

# METRIC CONVERSION FACTORS

## Approximate Conversions to Metric Measures

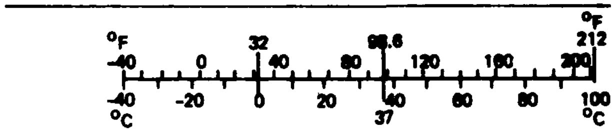
| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| in | inches | *2.5 | centimeters | cm |
| ft | feet | 30 | centimeters | cm |
| yd | yards | 0.9 | meters | m |
| mi | miles | 1.6 | kilometers | km |
| | | **AREA** | | |
| $in^2$ | square inches | 6.5 | square centimeters | $cm^2$ |
| $ft^2$ | square feet | 0.09 | square meters | $m^2$ |
| $yd^2$ | square yards | 0.8 | square meters | $m^2$ |
| $mi^2$ | square miles | 2.6 | square kilometers | $km^2$ |
| | acres | 0.4 | hectares | ha |
| | | **MASS (weight)** | | |
| oz | ounces | 28 | grams | g |
| lb | pounds | 0.45 | kilograms | kg |
| | short tons (2,000 lb) | 0.9 | tonnes | t |
| | | **VOLUME** | | |
| tsp | teaspoons | 5 | milliliters | ml |
| Tbsp | tablespoons | 15 | milliliters | ml |
| fl oz | fluid ounces | 30 | milliliters | ml |
| c | cups | 0.24 | liters | l |
| pt | pints | 0.47 | liters | l |
| qt | quarts | 0.95 | liters | l |
| gal | gallons | 3.8 | liters | l |
| $ft^3$ | cubic feet | 0.03 | cubic meters | $m^3$ |
| $yd^3$ | cubic yards | 0.76 | cubic meters | $m^3$ |
| | | **TEMPERATURE (exact)** | | |
| °F | Fahrenheit temperature | 5/9 (after subtracting 32) | Celsius temperature | °C |

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.




## Approximate Conversions from Metric Measures

| Symbol | When You Know | Multiply by | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| mm | millimeters | 0.04 | inches | in |
| cm | centimeters | 0.4 | inches | in |
| m | meters | 3.3 | feet | ft |
| m | meters | 1.1 | yards | yd |
| km | kilometers | 0.6 | miles | mi |
| | | **AREA** | | |
| $cm^2$ | square centimeters | 0.16 | square inches | $in^2$ |
| $m^2$ | square meters | 1.2 | square yards | $yd^2$ |
| $km^2$ | square kilometers | 0.4 | square miles | $mi^2$ |
| ha | hectares (10,000 $m^2$) | 2.5 | acres | |
| | | **MASS (weight)** | | |
| g | grams | 0.035 | ounces | oz |
| kg | kilograms | 2.2 | pounds | lb |
| t | tonnes (1,000 kg) | 1.1 | short tons | |
| | | **VOLUME** | | |
| ml | milliliters | 0.03 | fluid ounces | fl oz |
| l | liters | 2.1 | pints | pt |
| l | liters | 1.06 | quarts | qt |
| l | liters | 0.26 | gallons | gal |
| $m^3$ | cubic meters | 35 | cubic feet | $ft^3$ |
| $m^3$ | cubic meters | 1.3 | cubic yards | $yd^3$ |
| | | **TEMPERATURE (exact)** | | |
| °C | Celsius temperature | 9/5 (then add 32) | Fahrenheit temperature | °F |



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Frost formation of air-to-air heat pump evaporator surfaces reduces unit efficiency and restricts geographic application. The use of a fluidized bed heat exchanger as an air-to-heat pump evaporator was investigated to determine if frost accumulation could be eliminated. Experimental investigations were conducted and the following results obtained: (1) frost accumulation was insignificant with fluidized bed temperatures below 32°F and (2) moisture accumulation (continued)

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from condensation resulted in unstable fluidized bed operation when the fluidized bed temperature was above 32°F. Several concepts – maintenance of bed temperature below 32°F, reverse refrigerant flow, air-dry bed, nonadhering bed material, and ultrasonics – to solve the moist bed problem were evaluated, with no practical solution being developed. The use of a fluidized bed heat exchanger for air-to-air heat pump evaporators was determined not feasible.

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CONTENTS

## INTRODUCTION

One problem associated with air-to-air heat pumps is the accumulation of frost on evaporator heat exchanger surfaces. Such accumulations reduce heat pump operating efficiency by insulating heat exchanger surfaces and restricting heat exchanger air passages, resulting in the necessity for evaporator defrosting. Theoretically, a fluidized bed heat exchanger can eliminate the frost accumulation problem.

The Naval Civil Engineering Laboratory (NCEL) initiated an Independent Exploratory Development project to investigate the application of a fluidized bed heat exchanger as an evaporator for an air-to-air heat pump. The objectives of the investigation were to identify the conditions under which conventional air-to-air heat pumps experienced evaporator frost accumulations and then to determine if a fluidized bed heat exchanger could solve the frost problem.

Published sources provided the frost formation criteria needed to complete the first objective. A fluidized bed heat exchanger was then constructed, and its adaptability to heat pump applications was evaluated at NCEL. One characteristic observed - moisture accumulation from condensation within the fluidized bed - prevents adaptation of fluidized bed heat exchangers to air-to-air heat pumps.

## FROST ACCUMULATION WITH CONVENTIONAL AIR-TO-AIR HEAT PUMP

Frost accumulation with the air-to-air heat pump evaporator is dependent upon evaporator design, refrigerant temperature, ambient air temperature, and ambient relative humidity. The severity of frost formation increases with increases in the relative humidity and with decreases in ambient air temperature until slightly below 32°F. Below about 29°F, the rate of frost formation decreases.

The actual rate at which frost accumulates for any given ambient air temperature and humidity is dependent upon evaporator design and refrigerant temperature. For instance, an evaporator with closely spaced fins will accumulate frost at a different rate than one with wider fin spacing.

Evaporator frost formation rates are presented graphically in Figure 1 and are tabulated in Table 1. The frost formation rate data are based upon frost accumulation characteristics of several commercially produced heat pumps. Normalized frost formation rates were used to allow a comparison dependent only upon ambient air temperature and humidity. Typically, frost accumulations are sufficient to require evaporator defrosting from every 45 minutes (100% relative humidity at 29°F) to every 6 hours (70% relative humidity at 43°F).

Two methods for evaporator defrost are available: use of reversible units or use of nonreversible heat pumps. Reversible units defrost by reversing refrigerant flow in the system (Figure 2). The heat pump evaporator becomes a condenser and vice versa. During the defrost cycle, hot refrigerant vapor is pumped through the evaporator, and the frost accumulation is melted while the cold refrigerant is pumped through

the condenser. Heat is transferred from heated spaces to the outdoors for ice melting. Thus, to maintain the required level of heating during the defrost cycle, supplemental heaters, usually electric resistance type, are used to provide building heating and to replace the heat transferred for defrosting. Nonreversible heat pumps utilize evaporator heaters to remove frost. Several problems are associated with evaporator heaters such as decreased evaporator heat exchange efficiency and reliability; therefore, reversible cycle heat pumps are typically used.

## FLUIDIZED BED HEAT EXCHANGER

The fluidized bed heat exchanger is an approach based on the theory that if a fluid, at sufficient velocity, is passed through a bed of solid particles, the drag force on each particle will entrain it in the fluid flow. Under certain conditions, a bed of solids mixed with moving air, or some other fluid, can attain the physical properties of a fluid. The bed of solids, under these conditions, is known as a fluidized bed. Objects with densities less than the fluidized bed will float while objects with densities greater will sink. Waves can be formed and are propagated as they are in most fluids.

Heat transfer in a fluidized bed is based upon conduction and convection. For the heat pump evaporator application, heat is transferred to a tube containing a cold fluid (refrigerant) from the warmer fluid (air) used to fluidize the solid particles via the following:

- conduction - contact between the solid particles and the tube, or by particle contact with one another

- convection - by air passage over the tube or by air passage over the particles

One characteristic of a fluidized bed is that its temperature is essentially homogeneous; another is the large total surface area associated with the bed particles. Although the heat transferred by any one particle is small, the total heat transfer rate of a fluidized bed is great because of the large number of particles involved. Heat transfer coefficients of 10 to 200 times greater than for air-to-liquid heat exchangers are typical.

When a fluidized bed heat exchanger is used to transfer heat from the fluidization fluid (air) to a colder fluid within the bed (refrigerant) the following relationship exists:

$$T_a > T_b > T_r$$

where: $T_a$ = ambient outside air temperature

$T_b$ = fluidized bed temperature

$T_r$ = refrigerant temperature

A fluidized bed heat exchanger test prototype (Figure 3) was constructed at NCEL to provide data for application as an evaporator for an air-to-air heat pump. A medium pressure blower (2-1/2 inches of water) was used to fluidize a 1-1/4-inch bed of aluminum oxide particles whose diameters were less than 0.05 inch. A perforated brass plate with 1,479 holes/in.$^2$ (hole diameter of 0.016 inch) was used as an orifice plate. A finned tube heat exchanger with 1-inch fins (10 fins/in.) and a 3/8-inch-diam tube was used as the heat exchanger. Low outside air temperature was simulated by placing the fluidized bed heat exchanger test unit inside the NCEL cold test chamber. Refrigerant for the heat exchanger was supplied by a 1/3-hp R-12 compressor and condensing unit.

The fluidized bed blower requires more energy than required for a conventional air-to-air heat pump blower. Much of the additional energy is transformed to heat in the pressurized air and is recovered by the evaporator. Thus, the net energy consumption of the fluidized bed evaporator should be comparable to that of a conventional unit.

## FLUIDIZED BED HEAT PUMP EVAPORATOR EVALUATION

Two configurations of a fluidized bed evaporator were evaluated. The first evaporator configuration used a finned tube while the second used a bare tube. Frost formation on the evaporator, with and without the fluidized bed, was evaluated for each configuration.

The conventional finned tube evaporator experienced frost formation over a wide range of ambient conditions (60°F at 40% relative humidity to 23°F at 90% relative humidity). However, no frost accumulation was observed for the fluidized bed finned tube configuration when the fluidized bed temperature was less than 32°F. Operation of the fluidized bed evaporator with a fluidized bed temperature greater than 32°F resulted in moisture accumulation from water vapor condensation within the bed. The moisture caused the aluminum oxide bed particles to adhere to one another and form large lumps. This effect reduced the total particle-surface-area to weight ratio and the drag lift being provided by the air from the blower. Fluidization of the bed terminated and "blowouts" (free air channels through the bed) occurred. The stationary moist lumps of aluminum oxide in contact with the finned tube evaporator surfaces rapidly froze.

Frost accumulation for the bare tube evaporator was similar to that of the finned tube except that the frost formation rate was more rapid. This occurred because the heat exchange rate for the bare tube was less than the finned tube. Operation of the bare tube fluidized bed evaporator with a bed temperature greater than 32°F resulted in bed moisture accumulation, lumping, termination of particle movement, blowouts, and bed freezing. However, unlike the finned tube fluidized bed evaporator, the bare tube configuration had a thin layer of frost accumulating on the downstream side of the tube when the bed temperature was less than 32°F.

Problems in maintaining fluidized bed stability were experienced throughout the fluidized bed evaluation. These problems were based upon uneven air flow distribution and edge effects caused by heat exchanger surfaces and the outer walls of the fluidized bed container. The air distribution problem is common in fluidized beds but is solvable with

existing fluidized bed technology. The heat exchanger edge effects resulted from a large exchanger-dimension/bed-depth ratio and the use of a horizontal heat exchanger.

Figures 4 through 8 show the fluidized bed test unit operating in the NCEL cold test chamber: Figure 4, side view of the unit; Figure 5, end view of the unit; Figure 6, frost accumulation on a conventional finned heat exchanger; Figure 7, a fluidized bed in operation; and Figure 8, bare tube heat exchanger surfaces within the fluidized bed.

Several concepts were evaluated to attempt to solve the moist bed problem. Maintenance of a bed particle/air temperature below 32°F for all ambient conditions requires compressor operating pressures and evaporator control which are not practical. Reversing the refrigerant flow cycle to dry the moist bed provides no advantage over existing reverse-cycle air-to-air heat pumps. Termination of refrigerant flow with the air blower still fluidizing the particle bed will reduce the bed moisture content, but this consumes extra energy, is difficult to control, offers little or no advantage over existing heat pumps, and is not practical for ambient conditions having a high relative humidity.

Two other possible solutions to the bed moisture problem received only cursory evaluation: use of nonlumping bed materials or ultrasonics. Nonlumping bed materials were not actually evaluated. However, the addition of moisture to the fluidized bed increases the weight of material to be fluidized and the air pressure required for fluidization. The air pressure required for fluidization is dependent upon the bed weight. A variable pressure blower and a pressure control system would be required but such requirements will increase the cost and complexity of the fluidized bed exchanger to where it may not be competitive with conventional units.

The use of ultrasonics may provide a method to maintain moist bed stability. As in the previous concept a variable pressure blower and controls will be required. This concept also increases the cost and complexity of the fluidized bed evaporator. However, ultrasonics may provide an independent method to prevent frost accumulations on conventional air-to-air heat pump surfaces. An investigation of ultrasonics and air-to-air heat pump evaporator frost elimination is recommended.

## CONCLUSIONS

### Bed Stability

Fluidized bed instability occurred often and resulted from the moisture accumulations, uneven air distribution, and bed configuration.

Air distribution problems, which are caused by unlevel beds and unbalanced air passages, are correctible through design and air balance baffles. This is an engineering design problem and does not require research effort.

The bed configuration problem is normally a design problem for most fluidized bed applications. However, since this work is concerned with heat pump applications and, specifically, with improvement of heat pump efficiency, existing fluidized bed heat exchanger designs are not adequate.

The conventional design solution to the moisture accumulation problem would be to use a high pressure blower and vertical heat exchanger, but high pressure blowers are expensive and consume large amounts of energy. For successful adaptation of fluidized bed technology to air-to-air heat pumps, the total energy used for the fluidized bed exchanger must be low enough to justify the potential increased equipment cost. The use of low to medium pressure blowers reduces the amount of energy required and lowers equipment cost; however, the use of lower air pressures reduces the potential bed depths and requires the use of horizontal heat exchanger surfaces. The tube-diameter/bed-depth ratio used for the test unit was 0.375, which was too large and often induced blowouts. Further work is required to determine the ratio of optimum heat exchanger diameter to bed depth and particle size heat exchanger depth in the fluidized bed, and the ratio between the space between heat exchange tubes, tube depth, and particle size.

## Bed Moisture Accumulation

Fluidized bed moisture buildup occurred when bed particle temperatures were greater than 32°F. This resulted from water vapor condensation and produced negative effects. Damp aluminum oxide particles stick to each other, reducing the particle surface area and the drag lift provided by the air flow through the bed. It was found that eventually, particle movement stopped and the fluidized bed action terminated. Then blowouts occurred and the stationary damp aluminum oxide froze. No solution was found for the bed moisture accumulation problem.

## Fluidized Bed Air-to-Air Heat Pump

The adaptation of fluidized bed technology to air-to-air heat pump systems is not feasible because of the effects of bed moisture accumulation. Several potential methods to negate the moist bed effects were evaluated, but no practical solution was developed. Further effort related to the use of a fluidized bed to solve the air-to-air heat pump evaporator frost accumulation problem could not be justified and the project was terminated.

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Table 1. Heat Pump Frost Formation Rates[a]

| Temperature (°F) | Normalized Rate of Frost Formation at-- | | | |
|---|---|---|---|---|
| | 70% Relative Humidity | 80% Relative Humidity | 90% Relative Humidity | 100% Relative Humidity |
| 45 | -- | -- | -- | 0.30 |
| 44 | -- | 0.20 | 0.30 | -- |
| 43 | 0.7 | -- | -- | -- |
| 41 | 0.9 | 0.20 | 0.34 | 0.44 |
| 32 | 0.20 | 0.34 | 0.69 | 0.98 |
| 29 | -- | -- | 0.71 | 1.00 |
| 28 | 0.22 | -- | -- | -- |
| 26 | -- | 0.37 | -- | -- |
| 23 | 0.20 | 0.36 | 0.57 | 0.74 |
| 15 | 0.12 | -- | -- | -- |
| 14 | -- | 0.24 | 0.36 | 0.41 |

[a]Blanks indicate that data were not available at those temperatures.

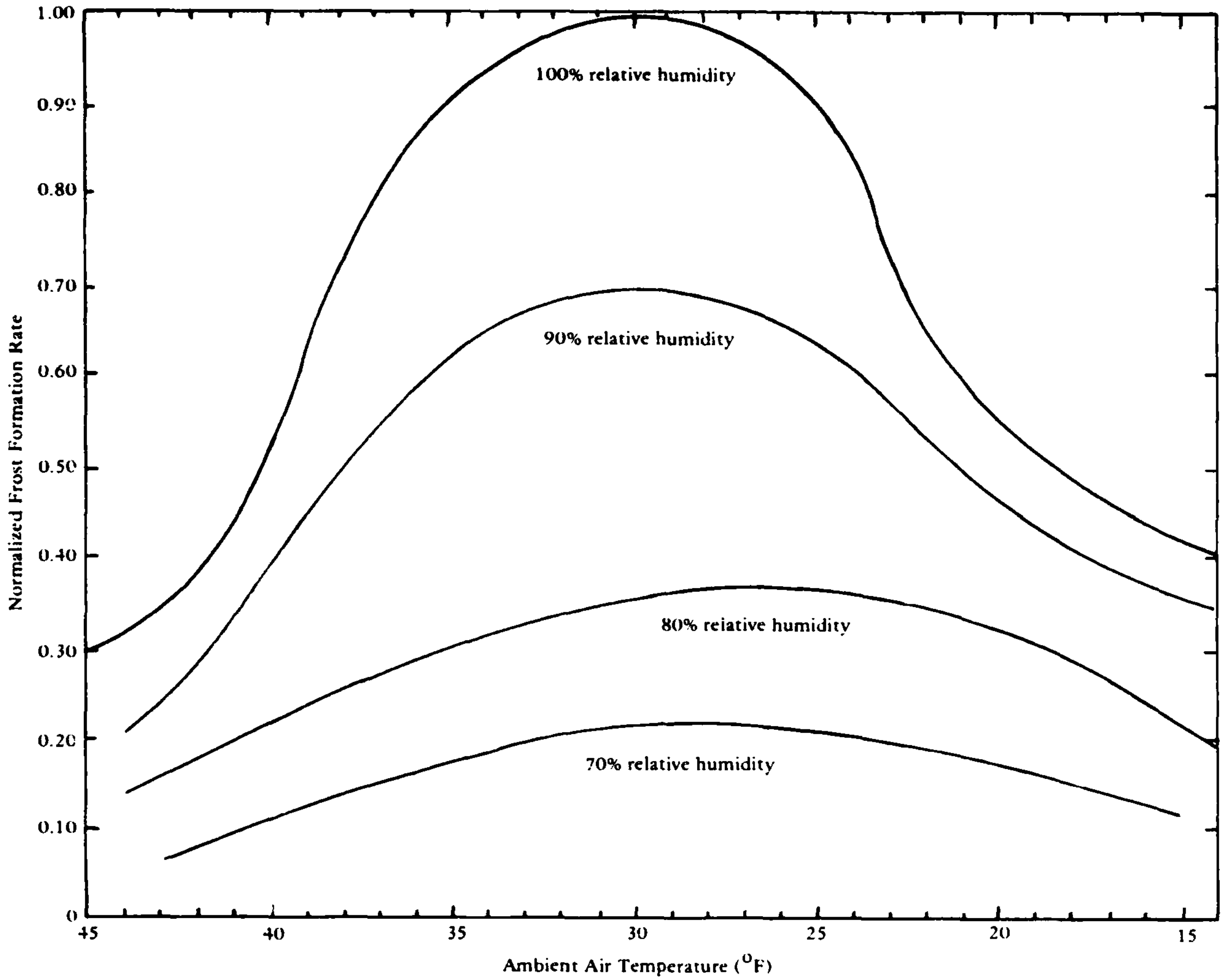
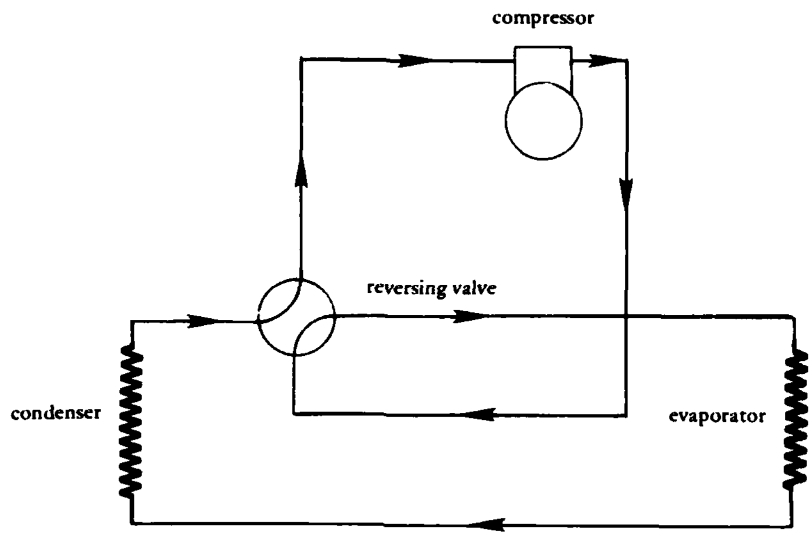
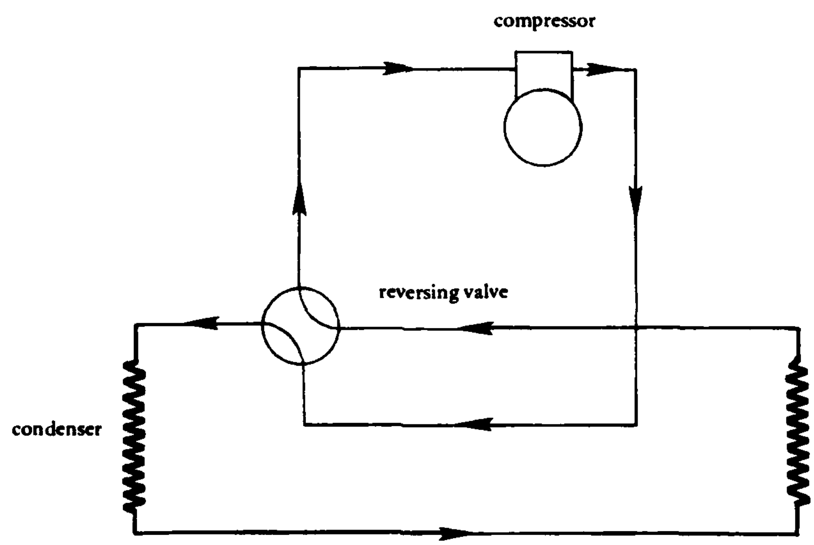



Figure 1. Air-to-air heat pump evaporator frost accumulation.




(a) Defrost cycle.




(b) Normal cycle.

Figure 2. Reversible heat pump.

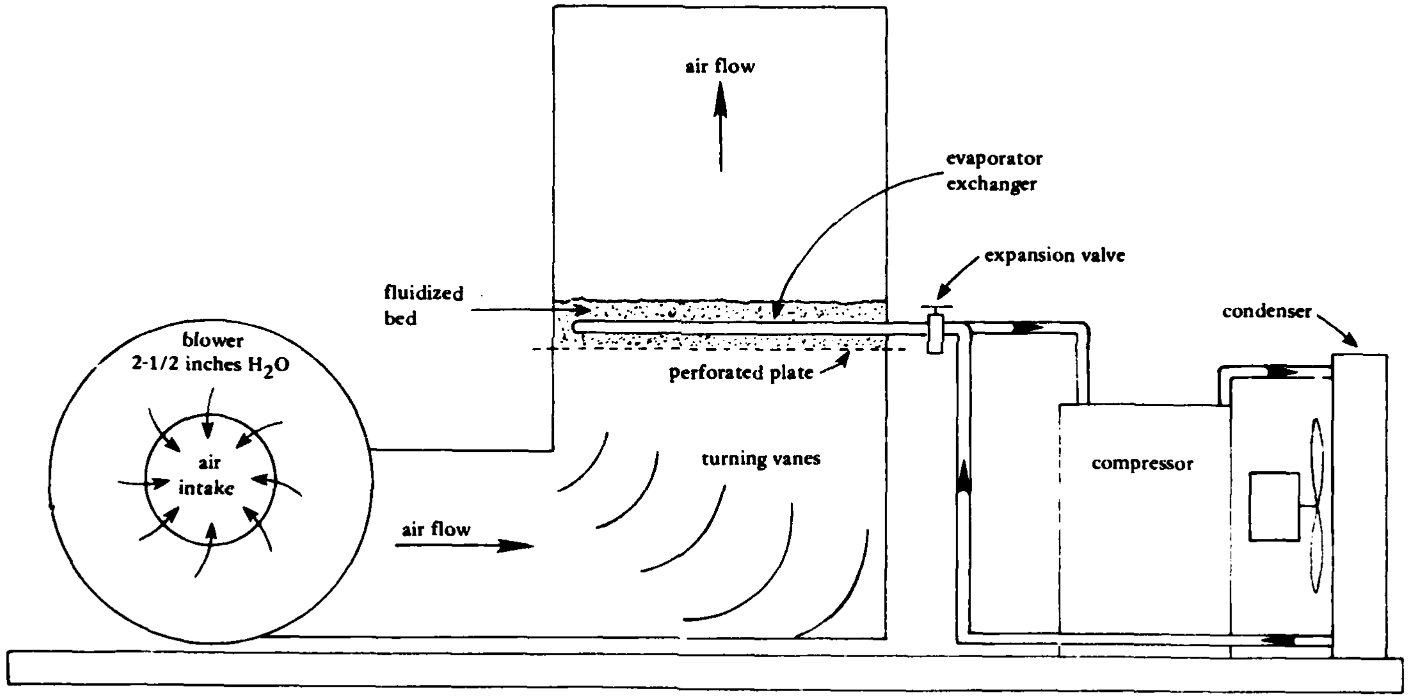



Figure 3. Fluidized bed test prototype.

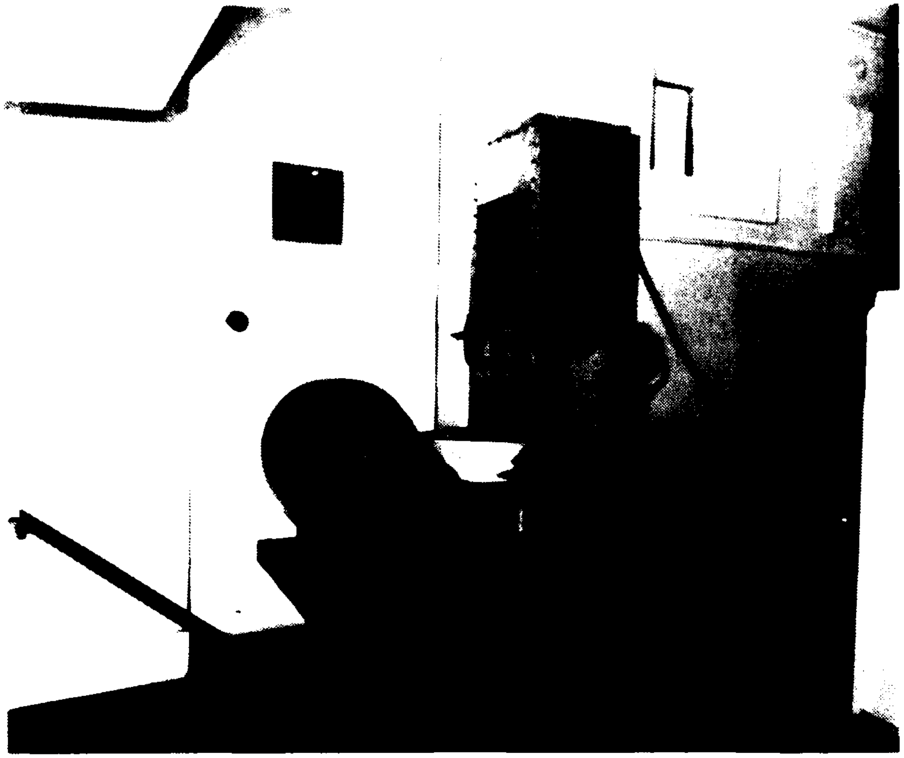


Figure 4. Fluidized bed test unit, side view located at the NCEL cold test chamber.

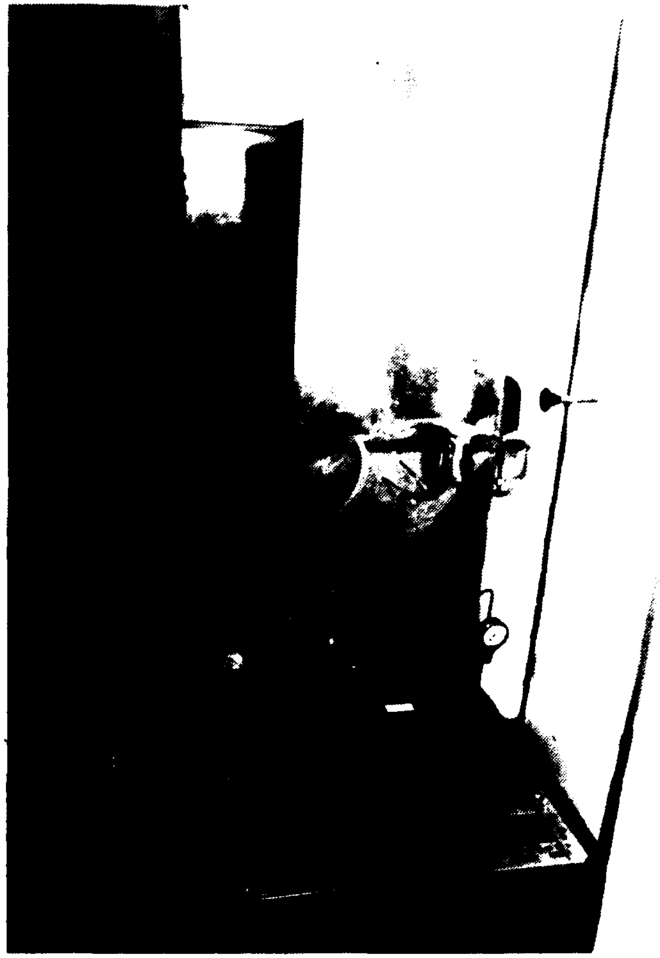


Figure 5. Fluidized bed test unit, end view, located at the NCEL cold test chamber.

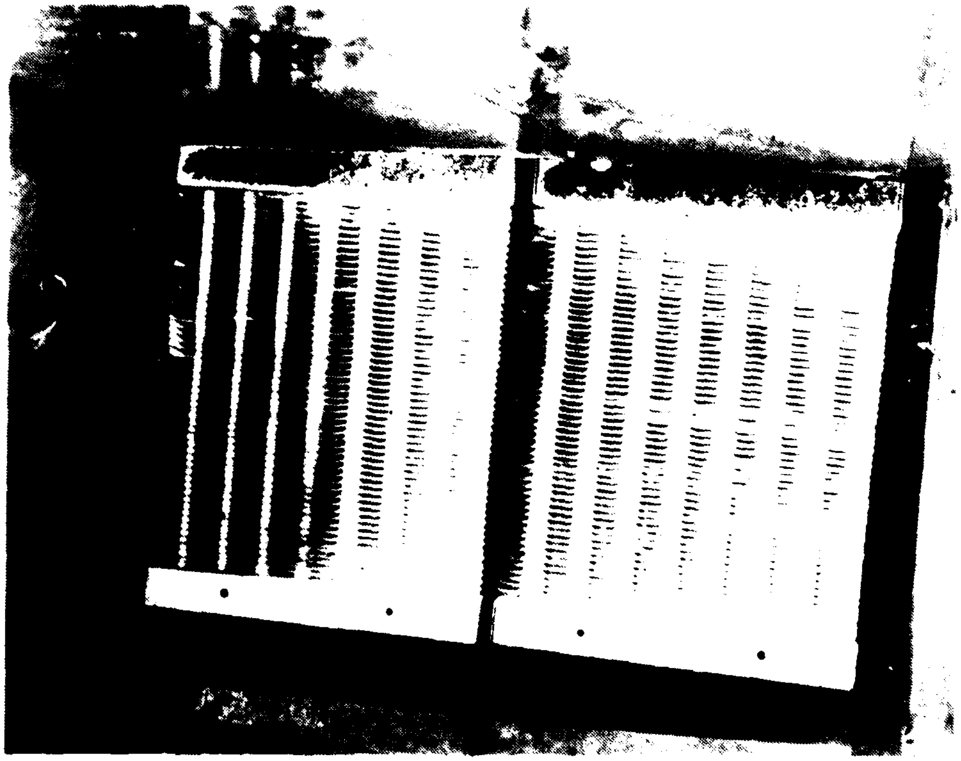


Figure 6. Frost accumulation, conventional finned tube evaporator.

Figure 7. Fluidized bed, top view.

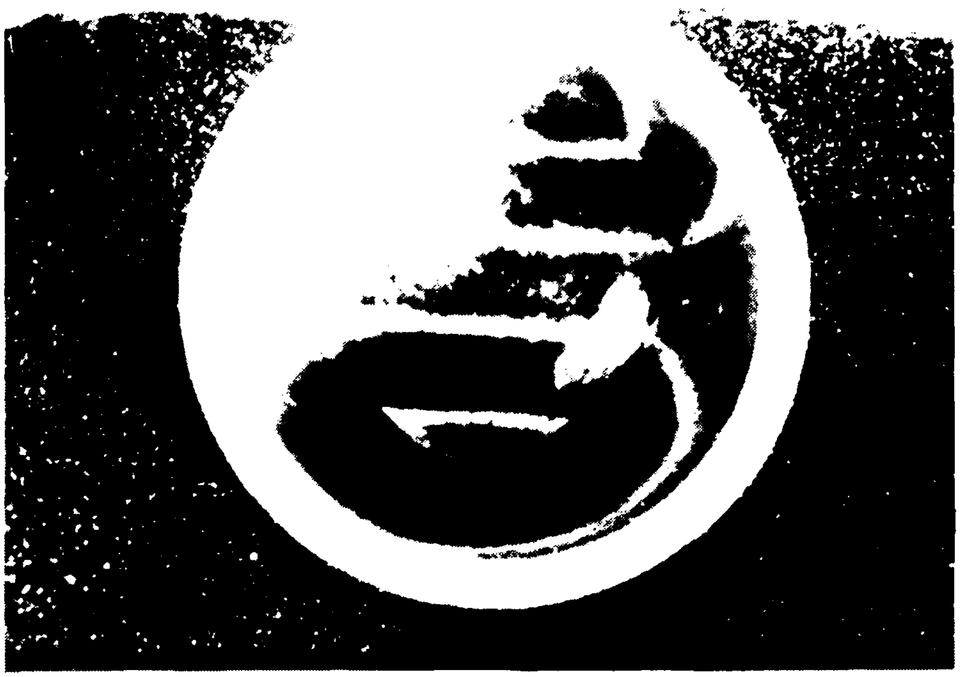


Figure 8. Fluidized bed, bare tube.

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MCAS Facil. Engr. Div. Cherry Point NC; CO. Kaneohe Bay HI; Code S4. Quantico VA; Facs Maint Dept - Operations Div. Cherry Point; PWD - Utilities Div. Iwakuni. Japan; PWO. Iwakuni. Japan; PWO. Yuma AZ
MCDEC NSAP REP. Quantico VA
MCLB B520. Barstow CA; Maintenance Officer. Barstow. CA; PWO. Barstow CA
MCRD SCE. San Diego CA
NAF PWD - Engr Div. Atsugi. Japan; PWO. Atsugi Japan
NALF OINC. San Diego. CA
NARF Code 100. Cherry Point. NC; Code 612. Jax. FL; Code 640. Pensacola FL; SCE Norfolk. VA
NAS CO. Guantanamo Bay Cuba; Code 114. Alameda CA; Code 183 (Fac. Plan BR MGR); Code 18700. Brunswick ME; Code 18U (ENS P.J. Hickey). Corpus Christi TX; Code 8E. Patuxent Riv.. MD; Dir of Engrng. PWD. Corpus Christi. TX; Dir. Util. Div.. Bermuda; Grover. PWD. Patuxent River. MD; Lakehurst. NJ; Lead. Chief. Petty Offr. PW/Self Help Div. Beeville TX; PW (J. Maguire). Corpus Christi TX; PWD - Engr Div Dir. Millington. TN; PWD - Engr Div. Gitmo. Cuba; PWD - Engr Div. Oak Harbor. WA; PWD Maint. Cont. Dir.. Fallon NV; PWD Maint. Div.. New Orleans. Belle Chasse LA; PWD. Code

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NROTC J.W. Stephenson, UC, Berkeley, CA
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PWC ACE Office Norfolk, VA; CO Norfolk, VA; CO, (Code 10), Oakland, CA; CO, Great Lakes IL; CO, Pearl Harbor HI; Code 10, Great Lakes, IL; Code 105 Oakland, CA; Code 110, Great Lakes, IL; Code 110, Oakland, CA; Code 120, Oakland CA; Code 154 (Library), Great Lakes, IL; Code 200, Great Lakes IL; Code 400, Great Lakes, IL; Code 400, Oakland, CA; Code 400, Pearl Harbor, HI; Code 400, San Diego, CA; Code 420, Great Lakes, IL; Code 420, Oakland, CA; Code 424, Norfolk, VA; Code 500 Norfolk, VA; Code 505A Oakland, CA; Code 600, Great Lakes, IL; Code 610, San Diego Ca; Code 700, Great Lakes, IL; Library, Code 120C, San Diego, CA; Library, Guam; Library, Norfolk, VA; Library, Pearl Harbor, HI; Library, Pensacola, FL; Library, Subic Bay, R.P.; Library, Yokosuka JA; Util Dept (R Pascua) Pearl Harbor, HI; Utilities Officer, Guam
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